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DESIGN AND ANALYSIS OF H-FRAME WITH LATERAL LINK SUSPENSION FOR AN OFF-ROAD VEHICLE

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ABSTRACT

The main objective of the paper is to design an H-frame suspension system for an off road vehicle by determining and analyzing the dynamics of the vehicle. The paper covers simulation, modeling and analysis of the suspension geometry. Suspension is designed such that it provides efficient dynamic conditions and enhanced stability of the vehicle.

KEYWORDS: H-Frame, FEA, Lateral Link, Off-Road Vehicle & Suspension

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Original Article

1. INTRODUCTION

A vehicle suspension system is a linkage to allow the wheel move relative to the body as when a tire hits an obstacle there is a reaction force originated which is reduced by the suspension of the vehicle to provide better handling and stability. The main objective of the suspension system is to maintain the contact between tire and the road. Some parameter affecting the suspension system is the ratio between the sprung and un-sprung weight as less the ratio more vehicle and body occupants will be affected by the bumps, dips and various other irregular surfaces. Another important parameter that has to be considered while designing the general suspension system of an off road vehicle is that the vehicle is made to over steer which in turn helps better control during corners. The H-frame suspension geometry is designed and analyzed for the better stability and enhanced the performance of an all terrain vehicle [1].

1.1. H-frame with Lateral Link

The H-frame with lateral link was selected for the rear suspension geometry due its lateral load carrying capacity in the lateral direction and the camber can be controlled through suspension travel. It consist of an h frame and the lateral link is also known as camber link because for controlling the camber and for carrying lateral loads.

The suspension systems consist of upper and lower lateral arms. The length of these links was obtained by using instantaneous center method for the better dynamic performance like minimum plunging of shafts. The upper arm is kept usually shorter to induce negative camber as the vehicle is in a turn, body roll results in positive camber gain on the outside wheel. The outside wheel also jounces and gain negative camber due to the shorter upper arm. These two effects are balanced so to keep the tire perpendicular to the ground especially for the outer tire because of the weight transfer during a turn. The hem joints are used for connecting the linkages and outboard ends of the arms is an upright with a spindle, hub which carries the wheel bearing and wheel [2-3].

1.2. Merits of H-Frame with Lateral Link

The following are the advantages of this suspension over the other suspensions systems [4-6]

- Enables more control over camber throughout the wheel travel
- The system has better anti –squat properties
- The problem related to plunging of the shafts can be minimized
- The force generated during the acceleration because of the weight transfer can be absorbed by the H frame by mounting it inclined backward
- The lateral links provide better load carrying capacity

2. DESIGN AND ANALYSIS

The important properties of a suspension related to dynamic are the kinematic behavior and its response to the forces and moments that it must transmit from the tires to the chassis. In addition, the main characteristics considered in the design process are manufacturability, weight, cost, assembly should be easy, space constraints.

2.1 Aim of a Suspension Design

- Reduce the un-sprung mass to have lesser inertia loads thus, the response time of the suspension changes in the track surface is decreased
- Increase in travel wheel travel to allow better absorption of ground shocks during the various obstacles
- Camber gain is provided while cornering for better handling and stability
- To minimize plunging of constant velocity joint which will further minimize the chances of popping out the half shaft from gearbox

2.2. Steps for Designing Suspension System

The steps for designing suspension are as follows [2, 6]

Step 1: The basic for designing a suspension system is to firstly decide the parameters through which the various points like roll center can be decided.

Step 2: By considering the parameters like camber, caster we can achieve better handling as during cornering negative camber gain should be there at the outer wheels and positive camber at the inner wheels for better stability.

Step 3: The roll center plays a very important role on a suspension steering response. So the roll center of the front suspension is kept lower than the rear suspension of the vehicle moreover, there is a direct correlation between roll center location and over steer, under steer, or neutral steer suspension behavior depicted in Figure 1.

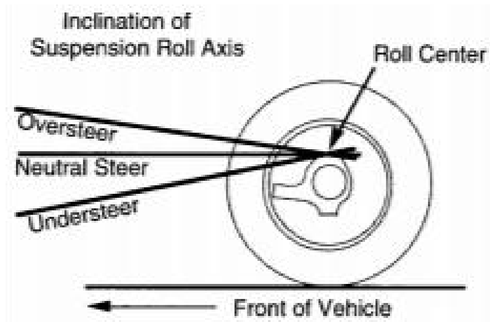


Figure 1: Roll Axis

Step 4: To start with the design firstly the vehicle main parameters such as wheel base, track width are decided then according to that suspension must be designed so that the tire edge surface is not more than the decided value.

The vehicle dimensions of off road vehicle are given in Table 1.

Table 1: Off Road Vehicle Dimensions

Vehicle dimensions	Rear view
Track width	48 inches
Wheel base	60 inches
Ride height	13.5 inches

Step 5: To keep a good distance from the road surface so to maximize obstacle avoidance 13.5 inches ride height from the lowermost member on the chassis to the ground was decided

Step 6: The track width was decided by taking in the consideration of the various parameters such as suspension travel and to aid in maneuverability. Wheel base was kept minimum to decrease the turning radius.

The recessional wheel travel is provided to account for the longitudinal forces that arise when a vehicle approaches a bump on the road.

Table 2: Suspension Parameters

Parameter	Rear Suspension
Suspension travel in jounce	6 inches
Suspension travel in droop	2 inches
Roll center height	9 inches

When the wheel approaches the bump or a pit on the road, it is subjected to vertical forces either (tensile or compressive) depending upon the load irregularity which are absorbed by the elastic compression, twisting properties of spring so to reduce the pitching tendency of the vehicle, the rear system should be more springing than the front system.

2.3. Design of H-Frame Geometry

The use of H-frame with camber link is special case where the H-frame is performing the function of four links instead of just three. It accomplishes this by reacting all braking load as torsional input. It is designed so that to minimize plunging of shafts. These lengths were obtained by the geometry using the instantaneous center method (Figure 2). In this the two links are extended to infinity where these two links meets at a point which is known as instantaneous center as it provide a better camber change during body roll [5]. The calculated values of the suspension design are given in the Table 3.

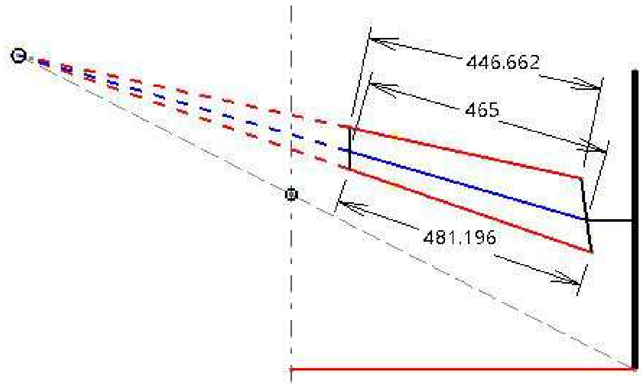


Figure 2: Determination of Link by Instantaneous Center Method

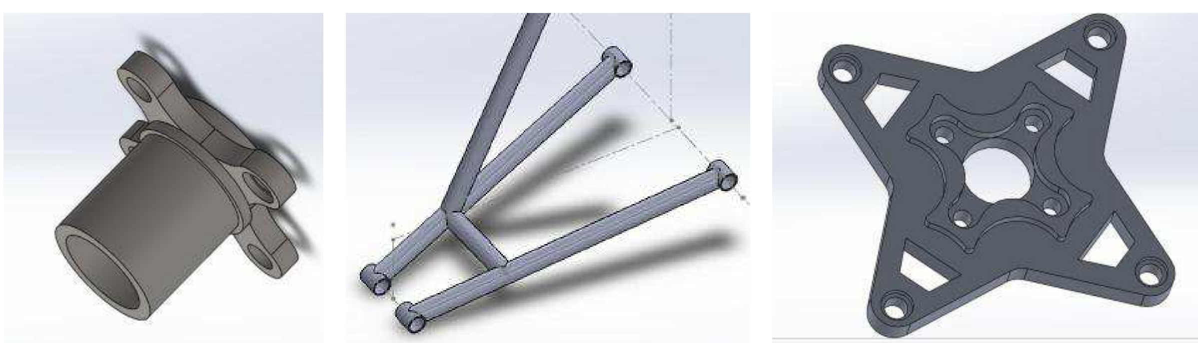
Table 3: Suspension Design Parameters

Parameter	Rear
Roll center	9
Motion ratio	0.7
Ride frequency	1.8 Hz
Wheel rate	6.37 N/mm
Ride rate	12.25
Spring rate	13.02
Roll rate	83.86
Camber change with travel	- 0.0323 degree/mm

3. MODELING OF THE SUSPENSION COMPONENTS

The following components were designed using Solid Works as shown in Figure 3(a,b,c).

- Rear hub (Figure 3a)
- Spindle (Figure 3b)
- H-frame (Figure 3c)



(a) **(b)** **(c)**

Figure 3: (a) Rear Hub, (b) Spindle and (c) H-Frame

3.1 Analysis of the Suspension

The ANSYS software was used for analysis of all the suspension components for combined loading of the forces. As per the ANSYS results, the specific changes were made to meet the specific decided objectives. The mainly three types

of forces were calculated for analyzing the suspension components. The three forces acting on the components were considered as follows.

- **Longitudinal Forces** – These forces arise due to braking and drive line forces acting on the components. The drive line forces which arises are generally been taken by the h frame and the breaking forces are carried by the hub with bearing carrier
- **Lateral Forces** – When a vehicle is taking turn a force equal to the centrifugal force is acted upon the contact patch of the wheel, thus a moment is induced in the suspension components and this coming load is been carried by the lateral links (camber links).
- **Vertical Forces** – These forces arise when a vehicle hits a bump. Usually 3G loads are acting on the components in case of bumps

3.2 Calculations for Determining Forces

3.2.1. Forces due to Reaction from the Ground

- Forces due to self weight:

$$F_1 = \frac{mg}{4} \quad (1)$$

- Dynamic forces on suspension components are considered to be 3 times the self weight:

$$F_2 = 3 \times F_1 \quad (2)$$

Considering weight of vehicle = 210 kg, F_2= 4855.5 N

3.2.2. Forces Arising while Cornering

During cornering centrifugal forces are acting on the vehicle and to keep the vehicle stable reactions are acting in the opposite direction at the contact patch of wheel, thus a moment is acting on the suspension components due to lateral forces.

Let,

M= moment acting on suspension components, Nm

R= radius of turn while cornering, m

r= radius of wheel, m

F_l=lateral force, N

F_c=centrifugal force, N

F_r=radial force, N

$$F_c = \frac{m \times v^2}{R} \quad (3)$$

F_c=3289.65 N

Considering a fraction of centrifugal force acting on rear wheels

F_r=1644.825 N

During cornering the weight of the vehicle is transferred to the outer wheels. Thus, one can assume that the centrifugal force is acting on the outer wheel,

Calculating the moments acting on the system:

$M = radial\ force\ \times radius\ of\ the\ wheel$

M = 459.56 N

3.3.3. Calculating the Longitudinal Forces

These forces arises due to braking action n of the vehicle

Breaking moment =159 Nm

The calculated forces acting on suspension system are given in Table 4.

Table 4: Calculated Three Forces Acting on the Components

Direction	Force
Longitudinal	159 N
Lateral	459.56 N
Vertical	3G N

These forces are applied on the various s suspension components through the ANSYS Software to analyze them. The Ansys results of H-frame and rear upright are shown in Figures 4, 5.

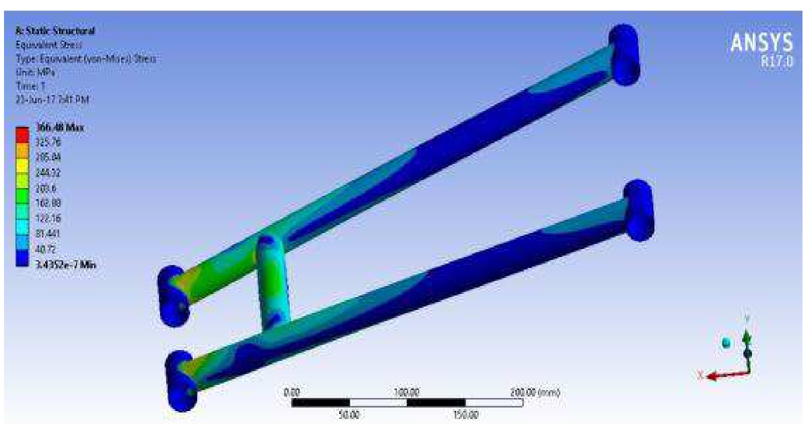


Figure 4: Analysis of H-Frame

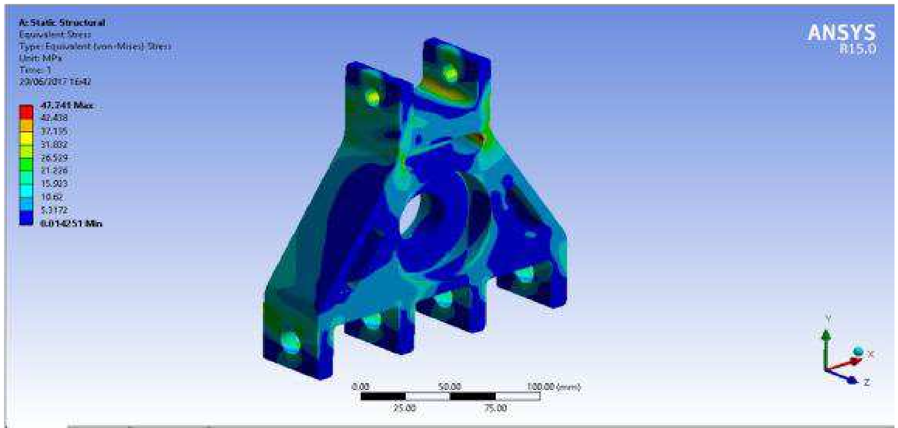


Figure 5: Analysis of Rear Upright (or Rear Knuckle)

3.3. Results of FEA Analysis of H-Frame and Rear Upright

The results of FEA analysis H-frame and rear upright are given in the Table 5.

Table 5: Results of FEA

Components	σ_{max} (MPa)	Factor of Safety
H-frame	366.48	1.81
Rear upright	47.741	6.912

3.4. Material Specification (Rear Knuckle, H-Frame)

3.4.1. Rear Knuckle

Material: AISI 6082-T6; Yield Strength: - 276 Mpa; Ultimate tensile strength:-310 Mpa

3.4.2. H- Frame

Material: AISI 4130

Table 6: AISI 4130 Specifications

Material	Alloy steel
Carbon	0.28-0.33
Cross Section	Circular
Diameter	1 ’’
Thickness	2mm

4. SPRING AND DAMPER SELECTION

Fox air suspension was selected which provide least adjustment of stiffness and progressive damping. Coil spring provides a constant stiffness which cannot be varied or changed. The mounting is difficult for this type of spring and it also increases the unsprung weight of the body. The stiffness of the air shock absorber can be easily varied by changing the air pressure in the cylinder pressure upto 150 psi can be varied. The selected shock absorber was FOX EVOL R (18.2’’) shown in Figure 6. Air spring curve is shown in Figure 7.

Figure 6: Fox Suspension

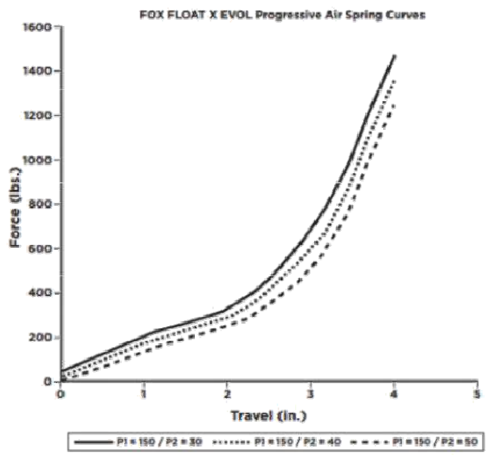


Figure 7: Air Spring Curve (Restoring Force Versus Travel)

5. SUSPENSION ANALYSIS USING LOTUS SHARK

Lotus Shark simulation software is used to verify the suspension design behavior dynamically. After designing the rear suspension system the hard points are imported from the CAD design and are inserted in the Lotus Shark simulation software as shown in Figure 8.

Rear Suspension Coords (3D)

	X (mm)	Y (mm)	Z (mm)
Point 1: Lower wishbone front pivot	170.5700	-115.5000	105.6000
Point 2: Lower wishbone rear pivot	-85.6000	-115.5000	105.6000
Point 3: Lower wishbone outer front pivot point	84.9200	-601.2700	0.0000
Point 5: Lower wishbone outer rear pivot point	0.0000	-601.2700	0.0000
Point 6: Upper link inner ball joint	-135.7900	-115.5000	212.3300
Point 7: Upper link outer ball joint	42.4500	-601.2700	121.4000
Point 8: Damper wishbone end	110.2700	-469.3200	5.5500
Point 9: Damper body end	42.4500	-300.6400	393.0100
Point 16: Upper spring pivot point	42.4500	-300.6400	393.0100
Point 17: Lower spring pivot point	110.2700	-469.3200	5.5500
Point 18: Wheel spindle point	42.4000	-601.2700	54.8000
Point 19: Wheel centre point	42.4000	-613.9700	54.8000
Point 20: Part 1 C of G	300.0000	0.0000	232.6000
Point 21: Part 2 C of G	3290.0000	500.0000	250.0000
Point 22: Part 3 C of G	3260.0000	665.0000	95.0000

Figure 8: Imported Co-Ordinates

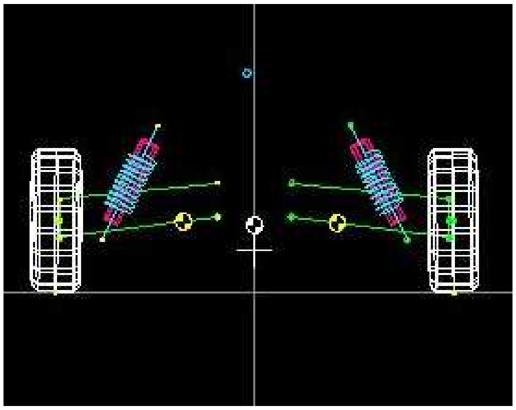


Figure 9: Front View of Rear Geometry

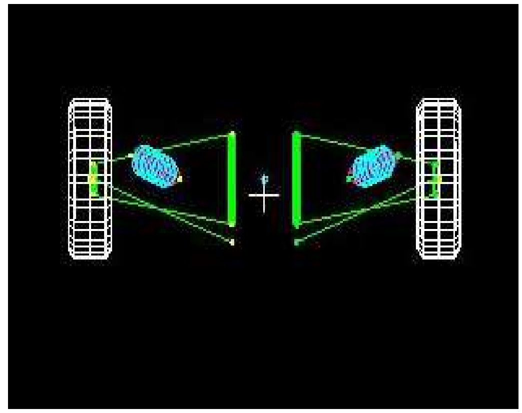


Figure 10: Top View of Rear Geometry

REAR SUSPENSION - BUMP TRAVEL
LHS WHEEL (-ve Y)

TYPE 2 H-frame Lower, single upper link

INCREMENTAL GEOMETRY VALUES

BUMP TRAVEL (mm)	CAMBER ANGLE (deg)	TOE ANGLE (deg)	CASTOR ANGLE (deg)	KINGPIN ANGLE (deg)	DAMPER RATIO [-]	SPRING RATIO [-]
-60.00	0.8797	0.0000			1.755	1.755
-40.00	0.5793	0.0000			1.750	1.750
-20.00	0.2867	0.0000			1.743	1.743
0.00	0.0000	0.0000			1.735	1.735
20.00	-0.2821	0.0000			1.725	1.725
40.00	-0.5611	0.0000			1.713	1.713
60.00	-0.8383	0.0000			1.700	1.700

INCREMENTAL SUSPENSION PARAMETER VALUES

ROLL ANGLE (deg)	ROLL CENTRE POSITION X (mm)	Y (mm)	Z (mm)	HALF TRACK CHANGE (mm)	WHEELBASE CHANGE (mm)	DAMPER TRAVEL (mm)	SPRING TRAVEL (mm)
	(+ve Y to outer wheel: Z rl to Grnd) (mm)						
-3.00	42.40	-32.73	158.37	-0.22	0.00	18.29	18.29
-2.50	42.40	-29.33	158.27	-0.16	0.00	15.27	15.27
-2.00	42.40	-22.32	158.76	-0.10	0.00	12.24	12.24
-1.50	42.40	-16.79	158.85	-0.06	0.00	9.20	9.20
-1.00	42.40	-11.41	158.84	-0.02	0.00	6.15	6.15
-0.50	42.40	-7.02	158.59	-0.01	0.00	3.08	3.08
0.00	42.40	-0.39	158.90	0.00	0.00	0.00	0.00
0.50	42.40	5.63	158.98	-0.01	0.00	-3.09	-3.09
1.00	42.40	9.03	158.66	-0.03	0.00	-6.19	-6.19
1.50	42.40	16.76	158.90	-0.06	0.00	-9.30	-9.30
2.00	42.40	22.15	158.88	-0.10	0.00	-12.43	-12.43
2.50	42.40	28.31	158.79	-0.16	0.00	-15.56	-15.56
3.00	42.40	34.40	158.27	-0.23	0.00	-18.70	-18.70

Figure 11: Simulation Result Sheet

The changes in parameters can be analyzed from the result of the simulated suspension geometry. The result shows the parameter change during bump travel and variation in roll angle. Changes can be made in the respective suspension geometry to obtain desired simulation result.

5. ASSEMBLED AND MANUFACTURED SUSPENSION COMPONENTS

The designed was validated by running and testing the vehicle in which the designed H- frame was used. While testing one problem was faced regarding the bending of the H-frame bracket. Further analysis was done to find the root cause of the problem. The problem was solved by proper alignment of the bushes and the brackets.

Figure 12: Assembled Rear Hub **Figure 13: H-Frame with Lateral Link**

6. CONCLUSIONS

- The entire H-frame geometry by simulated with the help of simulation software (lotus) and the hard points were verified. This helped in finding out the proper changes of the suspension parameters (camber) during the dynamic condition.
- By the proper simulation results it come to known that the tire remain perpendicular to the ground during the dynamic conditions like cornering which gives more stability and handling of the vehicle.
- By the proper design and manufacturing the problem of plunging of shaft was reduced.
- Thus the objective of rugged suspension system for an off–road vehicle was achieved.

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